

Gyroscopic Bearing Loads in Vehicular Flywheel-Based KERS

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ABSTRACT

In this work, we investigate the rotor bearing loads of a flywheel-based KERS that are caused by dynamic forces and gyroscopic torques during representative driving maneuvers. Based on the governing equations of motion of a gyroscope, the equations for the rotor-platform interactions are developed. These equations, which relate the vehicle's roll, pitch and yaw rate with the internal transverse torques on the flywheel, are integrated into a commercial vehicle dynamics program. An average passenger car model equipped with a typical high-speed flywheel energy storage system is used for the numerical investigations. The flywheel bearing loads produced by some selected, representative driving maneuvers are simulated for different orientations of the flywheel spin axis relative to the body frame. In addition, the dynamic response of the vehicle to the reaction torques is investigated in open and closed-loop vehicle dynamics simulations. Thus, the steering response of a driver model to the gyro dynamics of a flywheel-based KERS is obtained for fully charged KERS as well as during braking and under acceleration boosts. The result of the study is an overview of the flywheel bearing loading characteristics caused by vehicle maneuvering, and the vehicle's dynamic response to the induced gyroscopic torques.

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INTRODUCTION

Energy recuperation by regenerative braking has been in use extensively on railways for many decades and is recently finding its way into cars. The energy recovery mechanism slows the vehicle down by converting its kinetic energy into another form, which is then stored until it is needed for the acceleration of the vehicle. In electric railways the generated electricity is fed back into the supply system, whereas in battery electric and hybrid electric vehicles, the energy has to be stored in accumulators or ultracapacitors.

In the case of purely electric or hybrid electric vehicles, regenerative braking is easily implemented because they have already an electric machine that converts electric power into tractive power, or vice versa. For purely combustion engine powered vehicles an additional motor/generator unit would become necessary if the energy storage were provided via electrochemical batteries.

A more economical concept is to capture and store the mechanical or rotational kinetic energy of the wheels in a similar form, i.e., in the kinetic energy of a flywheel. In conjunction with a mechanical variator for transmission the losses associated with energy transformation can be minimized. In addition, a flywheel-based energy storage system can sustain a practically unlimited number of charging

cycles without reductions in energy storage capacity, whereas for electrochemical accumulators the number of charging cycles is limited.

Another competitive method, especially in the context of hybrid vehicles, is the use of electrical flywheel storage systems. Such a system consists of a motor/generator unit attached to the flywheel and additional motor/generator units, which are mechanically attached to the vehicle's transmission, driveshaft or wheels. Its capacity is higher than that of ultracapacitors and its power density is higher than that of Li-Ion batteries.

The use of flywheels for the storage of recuperated energy in automotive vehicles was reviewed in an SAE technical paper by R. C. Clerk as early as 1964 [1]. In 1979, A. A. Frank and N. H. Beachley discussed the problems emerging from the flywheels as energy storage systems in automotive applications, and emphasized the need for proper design [2]. The quest to improve the fuel economy of passenger cars in the aftermath of the oil crises has led to the investigation of an engine-flywheel hybrid powertrain at the General Motors Research Laboratories in 1984. An analytical estimation of the improvements that could be achieved in the fuel economy proved insufficient at that time to justify the complex drivetrain, and, therefore, a prototype system was never built (Schilke et al. [3]). Nowadays, according to the European Climate Change Programme, the vehicle fleets' CO₂ emission of every OEM must not exceed 95 g CO₂/km by 2020. This goal can only be

achieved by using hybrid drivetrains for middle and upper class cars, with an increasing field of application for flywheel storage systems.

Most important for the reemergence of flywheel-driven kinetic energy recovery systems (KERS) in automotive vehicles was the Fédération Internationale de l'Automobile's decision to authorize the use of hybrid drivetrains for the 2009 Formula 1 racing season. The specifications of the systems have been kept to a minimum with the clear objective of encouraging technical developments for use not only in motorsports but also mainstream automotive applications. A review of all the potential hybrid systems architectures led to the conclusion that the system which best met the specifications was a flywheel-based mechanical hybrid system. The UK-based companies Flybrid Systems LLP and Torotrak Ltd developed a system composed of a high-speed carbon filament flywheel enclosed in an evacuated housing and a full-toroidal traction drive continuously variable transmission (D. Cross and C. Brockbank [4]). In the same year two more UK companies, Williams Hybrid Power Limited (WHP) and Ricardo UK Ltd, developed flywheel units for Formula 1. The Ricardo concept features a flywheel in a hermetically-sealed vacuum vessel using magnetic gearing to couple the flywheel to a mechanical transmission, while the WHP flywheel is coupled to a motor/generator in an electrical powertrain. The flywheel's operating speed range of both Flybrid Systems and Ricardo is up to 60,000 rpm, supported by mechanical bearings, while the magnetized WHP rotor turns up to 36,000 rpm in hybrid ceramic ball bearings [5].

In motor sports applications the recuperated energy is rather used to improve acceleration than to increase fuel efficiency, although the new Formula One 2014 power train rules may pave the way for a fuel economy kinetic energy recovery system development for racing that may also be beneficial to road applications [6].

For the mainstream applications mechanical hybrid system are now in development for multiple automotive applications, and many passenger car manufacturers are examining the potential for implementation of such systems or have already done so. Several authors have investigated the potential energy saving benefits available with a flywheel-based kinetic energy recovery system and came to the conclusion that this technology may improve the fuel economy and hence reduce the CO₂ emission by more than 30 % in urban traffic (see e.g. [7]).

Another application that could evolve from such systems is rollover prevention, which employs the gyroscopic torques induced by the flywheel. Although the influence of gyroscopic effects on the dynamics of typical passenger cars is assumed to be almost negligible in standard driving maneuvers [8], the resulting gyroscopic torques of sufficiently dimensioned flywheels has been proved feasible for the active gyroscopic stabilization of articulated heavy-duty vehicles [9].

For the marketable development of flywheel-based KERS for vehicle applications radial and axial rotor support bearings are critical elements. These low-friction bearings must be able to withstand high speed, several sources of loads, have good vacuum compatibility as well as long life expectancies. These requirements frequently lead to choosing magnetic bearings for KERS applications, although complexity and costs are a major stimulus in the search for alternative solutions.

In the case of busses and heavy-duty vehicles, active or passive gimbal mountings are used to isolate the orientation of the flywheel rotor from the orientation of the vehicle chassis, thus allowing it to pitch and roll as freely as possible relative to the vehicle. To effectively decouple flywheel and vehicle yaw (which reduces gyroscopic loads on flywheel bearings during vehicle maneuvering), a restoring force is required to give the flywheel a home position, which is accomplished by springs or by gravity. Having a home position means the flywheel will precess, an unavoidable aspect of normal flywheel operation. This precession must be damped to minimize the probability of exceeding flywheel bearing loads and the allowable motion range of the support. The gimbal stiffness and damping parameters must be carefully chosen to avoid dynamic instability problems under certain vehicle curving conditions (see e.g. [10]).

For magnetic bearing systems the importance of a careful choice of elasticity and damping of the support is not to be underestimated even in non-gimbaled supports. In [11] it is shown that the housing dynamics can have a significant influence on the balancing of the rotor. Magnetic bearings provide relatively low support stiffness, which can result in flywheel precession that produces forces greater than the bearings are designed for, which can lead to severe touchdowns on the mechanical backup bearings.

In the simulations to follow, we have adopted flywheel dimensions which do not differ much from the Flybrid Systems specifications (see [4] and Table 1). Flywheel-based kinetic energy recovery systems of this type are recently in use not only in motorsports but increasingly in mass produced road vehicles. These systems consists of a single high-speed flywheel supported by mechanical bearings; rotating at over 60,000 rpm the flywheel can be very much smaller and lighter than has previously been possible, which allows the avoidance of softly suspended or gimbaled mounting system.

Optimal bearing design depends on the quantification of the bearing loads that are generated primarily by the vehicle's motion input at the mounting points of the flywheel. This motion can be classified as maneuvering (accelerating, braking, cornering), vibration (road excitation) and shocks (potholes). Another source of loads for flywheel bearings could be residual mass imbalance of the rotor. Due to precision balancing of the KERS flywheels this contribution is comparably small. The most significant sources of bearing loads are found to be due to road shock, and flywheel gyro dynamics [10].

In this paper, the flywheel bearing loads caused by the dynamic forces and gyroscopic torques produced by some selected, representative driving maneuvers are simulated for several alignments of the flywheel. The torques exerted by the flywheel in response to the rate of change of its angular momentum (i.e. its reaction torques) arise when the flywheel resists changes in the orientation of its spin axis caused by vehicle motion. The dynamic response of the vehicle to these reaction torques is investigated in several closed-loop vehicle dynamics simulations.

GYROSCOPIC TORQUES

When a vehicle containing a flywheel-based KERS maneuvers, some or all its rotation is imposed on the flywheel package. Vehicle maneuvering refers to the net rigid body motion of the vehicle. Corresponding linear acceleration of the vehicle leads directly to loads on the flywheel bearings. This would be the entire extent to which vehicle maneuvers need to be considered, were it not for the presence of gyroscopic action of the high speed rotor.

For the determination of the gyroscopic bearing loads caused by the vehicle's motion, the equation of motion of the flywheel has to be derived. This type of problem is quite common in aeronautics and spaceflight engineering, since attitude and stability control of spacecraft is usually accomplished by gyroscopes. We therefore follow the approach presented in [12, 13, 14, 15, 16] and take the rate of change of the flywheel's angular momentum $\mathbf{L}$ with respect to an inertial reference frame ($\mathbf{e}_1, \mathbf{e}_2, \mathbf{e}_3$), which equals the transverse torque $\mathbf{M}_r$ as the starting point:

$$\frac{d\mathbf{L}}{dt} = \frac{d}{dt} \mathbf{I}_r \boldsymbol{\omega}_r = \mathbf{M}_r, \quad (1)$$

with $\mathbf{I}_r$ and $\boldsymbol{\omega}_r$ being the flywheel's inertia tensor and angular velocity vector, respectively. The equation of motion needs to be expressed in the vehicle-fixed, non-inertial frame ($\mathbf{b}_1, \mathbf{b}_2, \mathbf{b}_3$, see Figure 1), which is accomplished by the well-known Euler's equations

$$\mathbf{I}_r \frac{d\boldsymbol{\omega}_r}{dt} + \tilde{\boldsymbol{\omega}} \mathbf{I}_r \boldsymbol{\omega}_r = \mathbf{M}_r, \quad (2)$$

with $\tilde{\boldsymbol{\omega}}$ being an skew-symmetric matrix representing the cross product with the vehicle's angular velocity vector:

$$\tilde{\boldsymbol{\omega}} = \begin{pmatrix} 0 & -\omega_3 & \omega_2 \\ \omega_3 & 0 & -\omega_1 \\ -\omega_2 & \omega_1 & 0 \end{pmatrix}. \quad (3)$$

The angular velocity of the flywheel is composed of the vehicle's angular velocity $\boldsymbol{\omega}$ (roll, pitch and yaw rate) and the wheel's relative spin ω_s :

$$\boldsymbol{\omega}_r = \boldsymbol{\omega} + \omega_s \mathbf{a}, \quad (4)$$

where $\mathbf{a}$ represents a unit vector in the direction of the flywheel's spin axis (see Figure 1).

In the flywheel's frame of reference, the angular momentum is simply given by

$$\mathbf{L}_s = I_s \omega_s \mathbf{a}. \quad (5)$$

In equation (5), the scalar quantity I_s represents the axial moment of inertia of the flywheel (the transverse inertia is denoted by I_t). In order to derive the net torque necessary to spin up the flywheel, its angular velocity has to be represented in the reference frame of the car, which is accomplished by projecting the vehicle's angular velocity onto the spin axis $\mathbf{a}$, and leads to

$$\mathbf{L}_a = I_s (\mathbf{a}^T \boldsymbol{\omega} + \omega_s) \mathbf{a}. \quad (6)$$

The torque propelling the flywheel, $\mathbf{M}_a$, is related to the rate of change of $\mathbf{L}_a$. From its magnitude the flywheel's angular acceleration can be derived as

$$\dot{\omega}_s = \frac{M_a}{I_s} - \mathbf{a}^T \dot{\boldsymbol{\omega}} \quad (7)$$

and substituted into equation (2), which delivers

$$\mathbf{M}_r = \mathbf{I}_r \left((\mathbf{1} - \mathbf{a} \mathbf{a}^T) \dot{\boldsymbol{\omega}} + \frac{M_a}{I_s} \mathbf{a} \right) + \tilde{\boldsymbol{\omega}} \mathbf{I}_r (\boldsymbol{\omega} + \omega_s \mathbf{a}). \quad (8)$$

In the above equation, $\mathbf{1}$ denotes the 3×3 unit matrix and $\boldsymbol{\omega}_r$ is substituted by equation (4). Equation (8) represents a coupled system of three non-linear, first-order differential equations, which has to be solved numerically.

The transverse torques $\mathbf{M}_r$ acting on the rotor will cause it to rotate about its center of mass. We simply state that each torque is caused by a couple of forces $\mathbf{F}_r$ on the rotor at a distance $l/2$ from the flywheel's center of mass, where l is the distance between the radial bearings. Here we assume that the bearings are located equidistantly from the center of mass. The superposition of the gyroscopic forces and the forces due to the linear and angular accelerations of the vehicle can be directly to the total loads on the flywheel bearings.

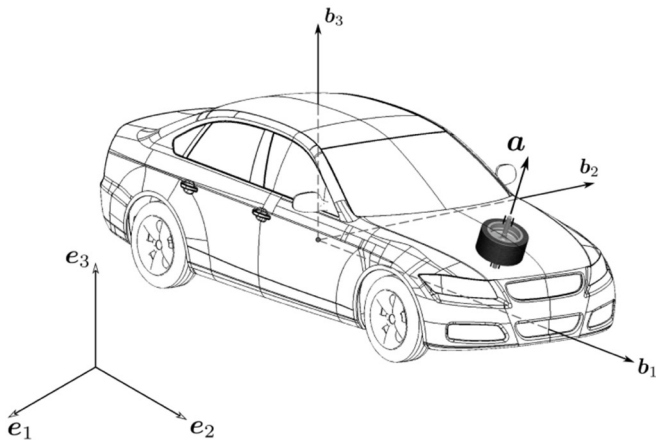


Figure 1. Schematic representation of the Vehicle-KERS system (composed of the DrivAer car model [17] and the Flybrid Carbon Fiber Flywheel [4]).

IMPLEMENTATION

The KERS is located above the rear axle under the trunk floor of the vehicle and is assumed to use the more familiar electrical powertrain with motors and generator, rather than a mechanical powertrain. The flywheel is solidly anchored to the chassis such that it must execute every turn experienced by the vehicle. In practice, of course, a certain flexibility of the mounting provides sound and vibration insulation. The elasticity and damping of the mounting must be chosen in such a way that the force transmission will be rather attenuated than amplified. The natural frequency of the support has to be adjusted sufficiently far away from the vehicle's pitch and roll natural frequencies as well as the resonance modes of the flywheel. In the following simulations we consider the system in the limit of rigid mounting in order to obtain a strong coupling with and a direct response of the vehicle. In this way we derive an upper limit of the bearing loads and of the vehicle's dynamic response to gross vehicle motion (maneuvering).

The highest angular velocities during normal operation of a road vehicle are experienced about the yaw axis. For this reason, it is generally thought that the best orientation of the flywheel axis is vertical, so that only the roll and pitch moments are present. Nevertheless, vehicle maneuvers result in motions about all three vehicle axes and traversing a speed bump, for instance, can produce higher angular speeds (pitch rates) of the vehicle's body than cornering. In order to evaluate the effects of the flywheel orientation, the following simulations of driving maneuvers are performed with different directions of the flywheel spin axis. In the case of a general orientation of the flywheel, **a** represents the unit vector in the direction of the flywheel's spin axis, and the vectors **b** and **c** are forming a right-handed orthonormal basis.

The net rigid body motion of the vehicle due to maneuvering, which is the primary contributor to bearing loads caused by gyroscopic torques, is simulated with the vehicle dynamics program veDYNA (TESIS DYNAware GmbH, Munich, Germany - see e.g. the overview article by Chucholowski et al.

[18] for more details). This program is based on a modeling concept that makes use of Jourdain's principle [19] and requires a MATLAB/Simulink (The MathWorks Inc., Natick, MA, USA) environment. It consists of a system of nine rigid bodies comprising the vehicle body, the axle suspensions and the wheels. Further sub-models are employed to depict the characteristics of the drive train, the steering mechanism, and the tires. Suitable minimum coordinates and generalized velocities are used to describe the spatial state of the vehicle and its components.

While the veDYNA vehicle model is coded in C, the assembly of the model components and the data flow are modeled in Simulink. The MATLAB/Simulink environment supports an open structure that allows external and user-defined models to be plugged in.

Table 1. Vehicle geometry and simulation parameters.

Vehicle mass	1324 kg
Wheel base	2.57 m
Wheel track	1.4 m
Center of Gravity (CoG) (*)	$(-1.25, 0, 0.25)^T$ m
Vehicle inertia at CoG	$I_{xx} = 305 \text{ kgm}^2$
	$I_{yy} = 1520 \text{ kgm}^2$
	$I_{zz} = 1750 \text{ kgm}^2$
Max. engine torque	224 Nm
Max. engine power	145 kW
Transmission gear ratio	1 st gear 3.830
	2 nd gear 2.199
	3 rd gear 1.401
	4 th gear 1.000
	5 th gear 0.810
Spring stiffness front / rear	17098 / 27183 N/m
Stabilizer stiffness front / rear	2610 / 197 Nm/°
Damping rate compression front / rear	1014 / 1064 Ns/m
Damping rate expansion front / rear	2400 / 2950 Ns/m
Aerodynamic drag ($A \cdot c_d$)	0.8 m ²
Tire size	205/60R13 88H
Max. longitudinal tire force (*)	3180 N
Longitudinal slip gradient (*)	70000 N
Max. lateral tire force (*)	3180 N
Lateral slip gradient (*)	35000 N
KERS maximum power	60 kW
Flywheel inertia	$I_s = 0.016 \text{ kgm}^2$
Max. flywheel speed	64500 rpm
Min. flywheel speed	15000 rpm
Distance between flywheel bearings	$l = 150 \text{ mm}$
KERS CoG position (*)	$(-2.75, 0, 0.25)^T$ m

(*) measured from center of front axle

(*) at 3000 N road load; degressive behavior is assumed

The vehicle model chosen for the simulation is the default sedan car offered by veDYNA. The essential geometry and simulation parameters used in the simulations are summarized in Table 1. The vehicle has a rear drive powertrain.

configuration with a manual transmission. The integrated KERS Simulink model transfers its power to and from the front wheels, which turns the default model into a hybrid car with regenerative braking.

The flywheel-based regenerative braking system operates in the three modes regenerative braking, acceleration boost and neutral, while the propulsion during constant speed phases is solely accomplished by the combustion engine.

The calculation of the gyroscopic reaction torque components M_{r1} , M_{r2} , M_{r3} , in [equation 8](#) as a function of the vehicle's roll, pitch and yaw rates (ω_1 , ω_2 , ω_3), angular accelerations ($\dot{\omega}_1$, $\dot{\omega}_2$, $\dot{\omega}_3$) and charging/discharging torque (M_a) is implemented in a MATLAB/Simulink function. It uses the motion data provided from the vehicle dynamics model and computes the gyroscopic torques. These torques, in turn, act back on the vehicle body. In addition, the function processes further data provided by the vehicle dynamics simulation, which are the accelerator and brake pedal position and the angular velocities of the front wheels. The braking strategy, which distributes the required braking force between regenerative braking and mechanical friction braking, gives priority to the kinetic energy recovery system acting on the front wheels. Proper amounts of frictional brake force on both the front and rear axle are applied to maintain stability and avoid wheel lock-up. For acceleration boosts the flywheel discharges its energy combined with the combustion engine. The magnitude of the applied torque is governed by the driver's acceleration request and by the wheel slip (veDYNA uses the semi-empirical TMeasy tire model [20] for the calculation of wheel slip). For further details, including a flow diagram of the embedded KERS subsystem in veDYNA, see [21].

VEHICLE DYNAMICS SIMULATIONS

In this chapter the loads on the flywheel bearings and the dynamic response of the vehicle to the KERS are investigated by numerical simulations of some standardized and non-standardized driving maneuvers. The use of the KERS is limited to acceleration and braking, while the propulsion during constant speed driving is exclusively carried out by the combustion engine.

The KERS is located above the rear axle under the trunk floor of the vehicle and the flywheel's axis of rotation is alternately aligned with the principal axes of the chassis in the simulations ($\mathbf{a} = \mathbf{b}_1$, $\mathbf{b}_2$, and $\mathbf{b}_3$ for longitudinal, lateral and vertical orientation of the spin axis, respectively).

Straight-Line Driving

In this maneuver the car accelerates for 20 seconds as fast as possible; it is then allowed to settle and the brakes are activated with 25 % of the available braking force. This relatively moderate braking power is chosen to prevent the wheels from locking when the KERS enters the regenerative braking mode. In this maneuver, bearing loads are produced

by linear acceleration due to pulling away, gear shifting and braking, and by the gyroscopic torque caused by pitch oscillations.

Vertical Flywheel Spin Axis ($\mathbf{a} = \mathbf{b}_3$)

The vehicle's pitch rate due to longitudinal acceleration is represented by the red line in [Figure 2](#). The moving-off of the car at second 2 causes a pitch rate of about $-10^\circ/\text{s}$; the gear changes take place approximately in seconds 3, 4, 7, and 13. The longitudinal load transfer incidental to the start of braking causes the car to pitch forward in second 22; its standstill produces the pitch oscillations in second 29.

The vehicle's pitch rate (ω_2) produces a gyroscopic torque about the $\mathbf{b}_1$ axis (M_{r2}), which is represented by the black line in [Figure 3](#). The first gear shift causes the highest torque (of about 30 Nm) due to the high rotational speed of the almost fully charged KERS.

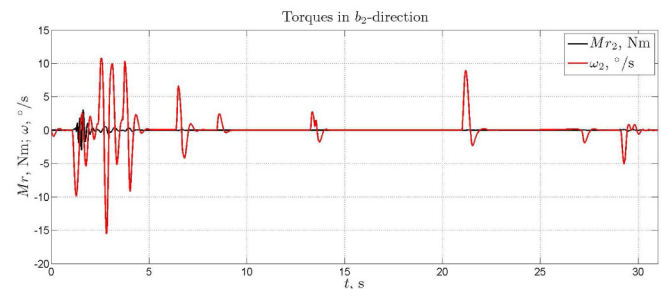


Figure 2. Closed-loop straight line driving maneuver with discharging/charging KERS. Torque M_{r2} and pitch rate ω_2 .

The shifting into the higher gears leads to smaller pitch oscillations and successively less gyroscopic torques. The gyroscopic torque caused by the vehicle's pitch motion produces a slight roll rate (ω_1), which, for its part, produces gyroscopic torque about the $\mathbf{b}_2$ axis (M_{r1}), as represented by the black line in [Figure 2](#). This torque, however, is more than an order of magnitude smaller than M_{r1} .

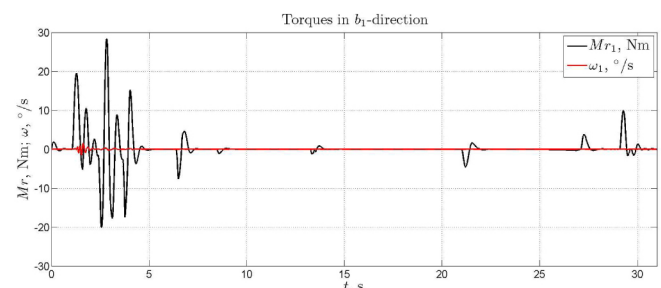


Figure 3. Closed-loop straight line driving maneuver with discharging/charging KERS. Torque M_{r1} and roll rate ω_1 .

[Figure 4](#) shows the torque M_{r3} , which is extracted from or fed into the flywheel. The increase during the acceleration phase corresponds to the decrease in angular velocity due to the power extraction from the KERS. During gear shifting the torque flow is interrupted, as depicted by the discontinuities. At the start of braking, the flywheel is at its minimum angular